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INTEGRAL PRIMARY AND SECONDARY HEAT EXCHANGER

TECHNICAL FIELD

This invention relates to structurally integrated condenser and secondary heat exchanger, and specifically to a structurally integrated condenser and oil cooler.

BACKGROUND OF THE INVENTION

An automobile requires several heat exchangers to dump waste heat to the ambient, including radiators to cool the engine, condensers to cool the air conditioning system refrigerant, and one or more secondary heat exchangers to cool secondary fluids such as engine oil and/or transmission fluid, and charge air coolers for super charging systems. Since ambient air flow is available through the vehicle grill, behind the grill has been the typical location for all such heat exchangers. In a continuing effort at cost reduction and simplification, and package size reduction, efforts have been made to structurally integrate as many components of these various front end heat exchangers as possible. For example, condensers and radiators have been combined into a front-back module, sharing a common tube header plate, as disclosed in USPN 5,509,199. Another and even older approach has been to package one or more heat exchangers in an over-under configuration. For example, USPN 2,037,845 shows an over/under charge air cooler and radiator, in which a series of parallel tubes extend between common manifold tanks, with the interior of the common manifold tanks divided into discrete sections for the two internal fluids (air and engine coolant) by double separators. While there is no particular discussion of tube thickness as measured in the direction perpendicular to external air flow (sometimes referred to as tube height), the tube thickness/height appears to be shown as common for the tubes in both the radiator and the charge air cooler sections.

Other patents disclosing over/under configurations for discrete heat exchangers recognize that a common tube thickness/height for the tubes in both heat exchangers is not necessarily optimal from a heat transfer standpoint. Oil, especially, is much thicker than engine coolant, and would require a larger flow passage. The internal height of the flow passage, together with tube wall thickness (determined by factors such as internal pressure resistance), determines total tube thickness/height. That is, overall tube thickness would be approximately the internal flow passage height plus twice the tube wall thickness. For fluids that are substantially less viscous than oil, the optimal internal flow passage height, and consequent overall tube thickness/height, would be substantially less. So, for example, USPN 4,923,001, which discloses an over/under engine cooling radiator and oil cooler, shows substantially thicker tubes for the oil cooler portion. Consequently, the oil cooler tube slots in the common slotted header plate must be thicker. This disparity in optimal tube thickness is even more pronounced when the primary heat exchanger (the one largest in face area) is a refrigerant condenser, and the secondary heat exchanger is an oil cooler. It is well known that the optimal internal flow passage height for a refrigerant condenser tube is on the order of .3-.6mm, see for, example, figure 3 of USPN 4,492,268, where the internal flow passage height is denoted at "d". This range of optimal internal flow passage height, even at the upper end, is substantially smaller than for oil, which may be two or three times more. Thus, in USPN 6,321,832, which shows an over/under configuration of condenser and oil cooler, it is recognized that the oil cooler tubes may have to be made thicker, and that the corresponding header slots that accommodate them would be thicker as well.

While thermal efficiency concerns dictate different thickness tubes when there is a disparity in optimal fluid passage internal height, manufacturing concerns dictate the opposite, since any lack of uniformity in component size and spacing detracts from manufacturing efficiency. Maintaining a common tube external envelope (common tube depth and thickness) lends itself well to a common header slot size and spacing, as well as a common air fin height. Thus, in USPN 6,394,176, an over/under combined

condenser and oil cooler is proposed which, just as USPN 2,037,845 discloses, shows a common header tank divided by a double separator, as well as a common tube outer envelope and air center height. The common tube outer envelope (called a common tube cross section) requires a compromise between the optimal flow passage size of the two tubes, a compromise that is quantified as an upper / lower range of the product of the hydraulic diameters of the two different flow passages. Since hydraulic diameter is a function of the flow passage cross-section and perimeter, it is essentially a function of the flow passage internal height, as well. In fact, for the typical square or round tube flow passage shape, the hydraulic diameter and internal flow passage height (one side of the square, or the diameter of the circle) are one and the same. In effect, it is the accepted wisdom that the maintenance of a common tube perimeter envelope or cross section will enforce a compromise in thermal performance of the respective tubes in the two operationally distinct (but structural integrated) heat exchangers. In the aforementioned patent, this compromise in condenser performance is manifested by the fact that the refrigerant flow pattern is a three pass pattern, indicative of a condenser flow passage size that is larger than the optimum, and optimum that would allow for a one pass design.

SUMMARY OF THE INVENTION

The integrated primary and secondary heat exchanger of the invention, which comprises a condenser and oil cooler, provides for a common tube exterior envelope, and specifically for a common tube thickness/height, but without compromising on the optimal interior flow height of the refrigerant flow passage.

In the embodiment disclosed, the oil cooler tube passage flow height is sized for its optimal performance, with a standard, single row, side by side array of flow passages along the width of the tube, thereby establishing the common tube thickness to be maintained. The condenser tube has its substantially smaller optimal internal height flow passages arrayed within the cross section of the condenser tube in a novel fashion that allows the use of and

optimally sized flow passage in both tubes. Rather than the standard, side by side array of flow passages, as in the oil cooler tube, the smaller sized refrigerant flow passages are arrayed along and around the perimeter of the tube, spaced from the exterior surface thereof by a minimal tube wall thickness that is established by conditions of tube material conductivity, strength, manufacturing capabilities, and other concerns. Depending on how much smaller the optimal refrigerant flow passages size is, compared to the oil cooler tube, a staggered array of passages may be created, or, ultimately, a double row of passages. The web of tube material between adjacent flow passages is maintained at at least the minimal thickness needed for burst pressure resistance. This more efficient packaging of the optimally sized refrigerant flow passages within a common size tube gives maximum assembly efficiency for the heat exchanger as a whole, without undue sacrifice in the thermal performance of the condenser.

BRIEF DESCRIPTION OF THE DRAWINGS

Figure 1 is a front face view of an integral condenser and oil cooler according to the invention.

Figure 2 is a cross section through an oil cooler tube.

Figure 3 is a cross section through one embodiment of a condenser tube made according to the invention.

Figure 4 is a cross section through another embodiment of a condenser tube made according to the invention.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Figure 1 shows a simplified plan view of the face of an integral condenser and oil cooler made according to the invention, indicated generally at 10. One section of 10, the primary heat exchanger, is a condenser, and is indicated generally at C. The other, significantly smaller, is beneath the condenser C, and indicated generally at O. That over/under configuration could be reversed, and depends only on where the refrigerant and oil lines run in any particular vehicle. The two sections of the heat exchanger maintain a high level

of part continuity and regularity. Thus, two common header tanks 12 and 14 are each with a regularly slotted header plate 16 and 18. These are numbered differently only to indicate opposite sides of the core, but would in fact be nearly identical parts. The condenser C has a series of condenser tubes 20, running in parallel from tank 12 to tank 14, and oil cooler O has a smaller number of oil cooler tubes 22, all with the same length (measured horizontally on the page, height/thickness (measured vertically on the page) and width/depth (measured perpendicular to the plane of the page). Heat exchanger 10 is the so called cross flow type, meaning that one fluid flow internally in one direction (right to left or left to right on the page), while air or some other external fluid flows perpendicular to that. Here, the external fluid is ambient air, but could be any common fluid. Brazed or otherwise conductively joined between the facing outer surfaces of the tube are a series of same height corrugated air fins or air centers 24. Ambient air flows over the air centers 24, perpendicular to the plane of the page, and over the exterior of all of the tubes 20 and 22, and centers 24 concurrently. In the process, the oil flowing inside of tubes 22, and the refrigerant inside of tubes 20, is cooled, and since the condenser is typically mounted in front of the engine cooling radiator, the cooling air is the "first in," unwarmed by any other heat exchanger. When the oil cooler is incorporated with the radiator, as is more common, then the air has typically already run through, and been warmed by, a conventional condenser. As disclosed, the interior of each header tank 12 and 14 is divided into discrete volumes to serve the condenser C and oil cooler O by double walled, or otherwise insulated, separators 26 and 28. The separators 26 and 28 allow the condenser C and oil cooler O to operate independently, and either tank 12 or 14, so divided, can serve as the inlet tank, or the outlet tank, for either one. As noted above, the insulated separators 26 and 28 are a well known means for dividing the common header tank. In summary, the high level of common parts, including the tanks 12 and 14, regularly slotted header plates 16 and 18, same exterior size tubes 20 and 22, and common height centers 24, maximizes assembly efficiency. As described next, this high degree of uniformity does not negatively affect the thermal efficiency of the condenser C.

Referring next to Figure 2, a cross section through an oil cooler tube 22 is shown. Oil cooler tube 22 is not novel in and of itself, but is combined with the novel condenser tube 20 as described below. Oil cooler tube 22 would typically be a unitary aluminum extrusion, with a width or depth D in the direction of airflow that is generally determined and limited just by the packaging space available at the front of the vehicle. As disclosed, D is approximately 18 mm. The overall shape is generally flat, but occasionally, and somewhat misleadingly, referred to as oval, because of the rounded leading and trailing edges. "Stadium shaped" is a more accurate term to describe the overall shape that is mostly flat, but with the edges rounded off. Inside of tube 22, a series of side by side flow passages, seven internal passages 30 and two edge tube edge passages 32, are surrounded on the outside by a perimeter wall 34 of thickness Two, and separated from one another by webs 36 with a thickness Wo. The actual values of Two and Wo are determined by several factors and limitations, including the necessary tube strength, (internal burst pressure within the passages 30 and 32), corrosion resistance of the wall 34, and basic manufacturing limitations on how thinly webs can be extruded. Here, Two is approximately 0.25 mm, and Wo is approximately 0.2 mm as well. The internal dimensions of the passages 30 and 32 are determined more by considerations of thermal performance, than by manufacturing limitations. This was not always the case, especially with the less advanced extrusion technology of a decade or two ago.

To understand the relationship between flow passage size and the refinement of extrusion technology over the last couple of decades, some brief explication of flow passage size in general is in order, as it relates to thermal efficiency. It has been a text book truism for many decades that the internal flow passages in a cross flow heat exchanger operate more efficiently as they progressively decrease in cross sectional area, all other things being equal. Quite simply, as the cross sectional area of the passage decreases, so the internally flowing fluid presents relatively more perimeter surface area to the externally flowing air, compared to the internal volume enclosed by the perimeter, and thus cools (or heats) more readily. This is intuitively obvious, in

the sense that a thin heated object will cool far more quickly in an air stream than will a thick heated object. In order to relate passages of differing cross sectional shapes (triangle, star shaped, etc) to the simplest shape, which is circular, the concept of "hydraulic diameter" was developed, which is a function of the ratio of cross sectional area of the flow passage to its perimeter, the units being length squared divided by length, or just length. As tube passage cross sections grow smaller, that ratio decreases, since relatively less area is enclosed by the perimeter. This is generally stated in text books as heat transfer increasing as the hydraulic diameter decreases. For simple passage shapes such as square or circular, the hydraulic diameter is simply stated as just the side of square or the diameter of the circle. For any design case, an optimal hydraulic diameter can be calculated, taking into account the fluid characteristics, the mass flow rate, tube length, tolerable pressure drops, and similar factors.

In decades past, extrusion technology had not advanced to the point where tube passages could be formed with an optimally small size. Here, as indicated in Figure 2, the optimal internal passage flow height Ho (equal to the hydraulic diameter in the case of a square passage) is approximately 1.5 mm, based on the flow characteristics of the oil, the mass flow rate, tube length, etc. That size passage can be produced by current, and even past, extrusion technology, and yields an overall tube thickness Tto, of approximately 2 mm (two times Two of 0.25 mm plus Ho of 1.5 mm). That will vary for other particular cases, but it establishes an overall tube thickness that it is desired to commonize over the entire core 10

As noted above, empirical tests had long ago determined that, for condensing refrigerant, the optimal internal flow height (for a simple shape passage) was in the range of 0.3-0.6 mm. This optimally small size could only be practically provided in a fabricated tube 20 or 30 years ago (such as separate webs brazed between upper and lower plates), but now extrusion technology has advanced to the point where extruded tubes can provide the same small passage size and short internal flow height. Other constraints, as noted above, still dictate a minimal perimeter wall thickness, which, in this case, is about 0.31 mm, and that value, plus the optimal internal flow height, together pre

determine a minimum overall tube thickness or height for a condenser tube. A minimum web thickness between flow passages that will withstand typical burst pressures in an aluminum tube is 0.2 mm, which does not affect overall tube thickness, but is still a design constraint. In the typical cross flow heat exchanger for automotive application, the designer will want to use the minimum overall tube thickness, and no more, so as to reduce air pressure drop and resistance across the core, as well as tube weight and cost. But, since the optimal internal flow height for refrigerant will typically be so much smaller for refrigerant than for oil, a design decision to standardize the total tube thickness for both the oil cooler and condenser tube requires a condenser tube that is thicker than it would ideally be in a condenser alone. Stated differently, the common total tube thickness, minus the smaller internal flow height of the refrigerant passage, yields a tube perimeter wall thickness far greater than the minimum needed.

Referring next to Figure 3, the design according to the invention of a tube for a primary heat exchanger tube in general is illustrated. The terms primary and secondary are arbitrary, and the specific case of a condenser tube as the primary tube is consider below. A primary tube 50 is illustrated in which the optimum internal passage flow height H is substantially less than for the oil cooler tube 22 (or other secondary tube), though not as small as it typically would be for a condenser tube. Tube 50 has the same depth D and total thickness T equal to that of oil cooler tube 22. Stated differently, its exterior or outer envelope is uniform. A minimum perimeter wall thickness Tw and and web thickness W as illustrated are comparable to oil cooler tube 22. Round internal flow passages 38 are shown which, as noted above, have a diameter equal by definition to the hydraulic diameter and also equal to the internal passage flow height H. Since the secondary tube, such as the oil cooler tube 22, determines the total overall tube thickness T, placing the substantially smaller flow passages 38 in a standard, side by side array would give a perimeter wall thickness that was substantially greater than the minimum needed. Instead, according to the invention, the flow passages 38 are arrayed around the perimeter of the tube 50, always within the minimum wall thickness Tw from

the perimeter, and always no closer to an adjacent flow passage 38 than the minimum web thickness W. For the case shown in Figure 3, this effectively creates a perimeter wall 40 which, while it does not have a constant wall thickness, is no thinner than the minimum thickness Tw at any point. Likewise, the flow passages 38 are separated from adjacent flow passages 38 by webs 42 that are no thinner than the minimum web thickness W at any point.. So, all the requirements of tube strength are met. While the flow passage array is odd looking by conventional standards, more flow passages 38 are packaged within the available envelope than would be the case with a conventional, side by side array, and with a perimeter wall 40 which, on average, is thinner than the overly thick perimeter wall that would be created by such a conventional array, so that weight and cost are reduced.

Figure 4 illustrates the case when the primary tube is the condenser tube 20, with its even smaller optimum internal flow height Hc of approximately 0.5 mm, which is the same as both the passage geometric diameter and its hydraulic diameter. This optimum Hc is less than half that of the oil cooler tube 22, and small enough to allow for a one pass flow pattern through all condenser tubes 20. This indicates that no compromise in the thermal performance of the condenser tube 20 or its value of Hc was made in order to keep its total thickness equal to the oil cooler tube 22. A minimum wall thickness Twc of 0.31 mm is maintained, and a minimum web thickness Wc of 0.2 mm. Far more flow passages can be packaged, creating an even thinner (on average) perimeter wall 46 that with tube 50, and webs 48 of adequate thickness. In effect, by following the design constraint described, a double row of such flow passages 44 is created, not just a staggered row, two rows of 23 in this specific case, with an extra end passage placed near each rounded tube edge, for a total of 48. Only about half that number of flow passages could be accommodated with a conventional, side-by-side array. While the flow passages 44 are separated from each other, in the direction normal to air flow, by significantly more than the minimum web thickness needed (approximately 0.4 mm in this case), there is not sufficient room for the addition of any extra flow passages between the two rows. If, for some reason, the optimum Hc were

significantly less than illustrated (a different refrigerant, a far lower mass flow rate, a shorter length tube, etc), then there might be sufficient room between the rows. Such extra flow passages would not be especially efficient, thermally speaking, being barricaded from the perimeter of the tube 20 by the perimeter passages 44, but might make sense in some cases, to reduce tube weight, if nothing else. In any case, there will be at least the array of flow passages around the perimeter of the tube.

In conclusion, a condenser tube designed for use in a typical, refrigerant only condenser core would, with flow passages of the size shown in Figures 3 and 4, have much thinner tubes. For example, in Figure 4, with Hc equal to 0.5 mm, and Twc equal to 0.31 mm, total tube thickness would be only about 1.12 mm, almost half as thick, with the flow passages arrayed side by side, if the tube were designed for a condenser alone. To array the flow passages any other way, and thereby increase the total tube thickness Ttc, would be counter intuitive and counter productive, because of the added air pressure loss. But, with the added manufacturing design constraint of Ttc being equal to the thicker Two, the odd appearing flow passage packaging scheme of the invention provides an advantage, by packaging much more refrigerant flow area within the same envelope, and reducing added metal, weight and cost to the tube. Furthermore, the thermal performance of the two tubes 20 and 22 is not compromised to any great extent, as it is with conventional, side by side flow passage arrangements, as are found in other integrated, primary and secondary heat exchangers. That is, the hydraulic diameter/internal flow passage height is optimized for both the oil and condenser tubes, independently of each other, and not limited to a compromise product of the two variables that inevitably requires one variable to be too large as the other approaches its optimum, and vice versa.

Variations in the disclosed embodiments could be made. As noted, optimal values for the internal flow passage height will vary with the fluid characteristics, flow rates, total tube lengths and tolerable pressure losses. In general, for a combination of oil cooling and refrigerant condensing, and with a typical tube length found in a vehicle front end heat exchanger 500 to 900 mm), a range of 1 to 3 mm for the internal passage flow height in the oil cooler

tubes 22, coupled with a range of 0.3 to 1.0 mm for the condenser tubes 20 will allow for independent, thermally efficient operation, with acceptable pressure drops, while maintaining a common structural foot print. Again, this is tantamount to establishing a range of hydraulic diameter for the flow passages, for the case of the simple flow passage shapes involved. Flow passage shapes other than simple round or square could be used, such as a square shape enhanced by internal fins, a common way to increase effective flow passage perimeter surface (and thereby decrease hydraulic diameter) in the past, before extrusion technology had advanced to the point where simple shapes could be extruded with a small enough internal flow height to inherently have an optimally small hydraulic diameter. Such a simple shape (round, square, rectangular), with a simple internal flow height, is preferable with today's technology. Regardless, even a finned or otherwise complex flow passage shape will have an average or de facto internal flow height, so that optimizing a flow passage hydraulic diameter is essentially the same as optimizing the internal flow passage height, which is a simpler quantity to deal with conceptually, especially in terms of total tube thickness. The header tanks 12 and 14 need not be common tanks, as shown, and might not be made as common tanks, if cross contamination or thermal cross conduction between the two fluids were a great concern. In other words, four separate header thanks could be used, with no internal separators, but a common header plate regularly slotted and with standard tube and center sizes, could still be used, even if the tanks were separate for each of the two heat exchanger sections. One pair of opposed slots could be left empty, and one tube removed, so as to the adjacent ends of such separate tanks. Such a design would still maintain almost all of the manufacturing advantages of common component size.